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Theoretical and experimental investigation of diesel engine with steam injection system on performance and emission parameters

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HIGHLIGHTS

► Steam injection system is applied to DI diesel engine to reduce NO*x* emissions.
► This method improves performance and emissions as a result of experiments.
► Combustion model was used to estimate emission and performance parameters.
► Combustion model has a quite well agreement with the experimental results.

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ABSTRACT

In the present study, a new electronically controlled steam injection method is applied to a direct injection (DI) diesel engine to control NO*x* emissions. This method can be also used to improve the performance and efficiency. Steam injected diesel engine is modeled by using zero-dimensional single-zone combustion model for 20% steam ratio at full load condition. The obtained results are compared with conventional diesel engine in terms of performance and NO, CO, CO_2, HC emissions. The simulation results agree with experimental data quite well. In the experimental results, it is determined that the engine torque and the effective power increase up to 2.5% at 1200 rpm, specific fuel consumption (SFC) and effective efficiency improves up to 6.1% at 2400 rpm, NO emissions reduce up to 22.4% at 1200 rpm, CO_2 emissions decrease up to 4.3% at 1800 rpm, smoke density increases from 44% to 46% at 2200 rpm. This paper may be a leading essential tool for the real-engine designers by considering the effects of steam injection into the engine cylinder.

1. Introduction

There are many negative effects of emissions released from sea and land vehicles on human beings and environment. Because of these detrimental effects, emissions released from diesel engines are gradually limited by regulations [1].

There are various methods inside and outside of the cylinder to reduce NO*x* emissions [2–5]. One of the NO*x* reduction methods is water injection into the engine cylinder with different methods. The water is injected to combustion chamber as water–fuel emulsion or directly and also it could be injected into intake manifold (Fumigation). In the literature, it is observed that NO*x* emissions can be reduced using water since maximum flame temperature decreases [4,5].

Armas et al. expressed that combustion process would be much faster [6], hence effective efficiency increases and adiabatic flame temperature decreases [7], the formation of soot, NO, HC and PM emissions reduces [9] with respect to the increased amount of OH radicals [8] because of water dissociation when using emulsified fuel in diesel engine. Samec et al. emphasized that NO*x* and soot emissions reduce 20% and 50% using 10% and 15% water/fuel emulsion, respectively. On the other hand, there is no an increment in the SFC [4]. Abu-Zaid et al. observed evaluation in torque, power, effective efficiency, SFC and exhaust temperature as water/fuel ratio increases [10]. Lin and Wang indicated that exhaust temperature, O_2, NO*x* and smoke decrease, CO_2 and CO emissions increase when three phase emulsified fuel (water/fuel/water or fuel/water/fuel) injected into diesel engine and three phase emulsified fuel has higher exhaust gas temperature and lower CO and NO*x* emissions when compared to two phase (water/fuel) [11]. In the study of Alahmer et al., SFC and CO_2 emissions increase; NO*x* emissions reduce by using 5–30% emulsified diesel fuel [12].

In fumigation method, water in liquid phase is injected into intake manifold. Tauzia et al. stated that whilst water 60–65% is

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injected into intake manifold of diesel engine with common rail injection system, NOx emissions reduce up to 50%, heat losses increase at cylinder walls, because of this effect, a reduction is seen in engine efficiency, and also in PM emissions considerably [13]. Donahue and Ishida et al. investigated water injection into diesel engine and observed that whilst NOx, soot emissions and SFC reduce at low loads, soot and SFC increase and NOx emissions decrease at high loads [14,15].

Another method is direct water injection into the combustion chamber. This method has an advantage according to fumigation method as the water is injected directly to the flame core [16]. Christensen et al. expressed that when water is injected to HCCI (Homogeneous charge compression ignition) engine; combustion rate decreases, unburnt HC and CO emissions increase, NOx emissions reduce [17]. Samec et al. investigated the injection of water into diesel engine during combustion. It is pointed out that NOx emission considerably reduces as a little reduction which is seen in soot emissions [18]. Bedford et al. performed a research on direct water injection into the diesel engine by means of Computational Fluid Dynamics (CFD) program and found that SFC, PM and NOx emissions reduce at 44% load, NOx emissions reduce and only a little increment in SFC at 86% load [5].

As can be seen above literature review, water is injected into combustion chamber directly, as emulsified or fumigation into the intake manifold. As a result, it is observed that NOx emissions reduce [4,11,13], HC and CO emissions and SFC increase [10–12]. Besides, condensed water particularly in fumigation method deteriorates the specification of lubrication oil and it is stated that wear rate of moving part of engine increases consequently [19].

One of the proposed methods to decrease NOx emissions is water injection to intake manifold as a steam phase. Parlak et al. determined that NOx emissions reduce up to 33%, effective power and torque increases up to 3% and SFC decrease up to 5% as a result of full load tests with electronically controlled steam injection system. Furthermore, optimum steam ratio was determined as 20% (S20) [20,21]. Murthy et al. observed that steam injection which is solar generated into diesel engine, NO emissions and exhaust temperature reduce, soot emissions, power and SFC increase at full load conditions [22].

Based on previous literature reviews, there is no theoretical and experimental study together in order to investigate performance and NO emissions parameters of diesel engine with steam injection system Therefore, in this study, electronically controlled steam injection system developed by Parlak et al. [20] is used to investigate torque, effective power and efficiency, SFC, inside cylinder pressure and temperature, NO, CO, CO_2, HC and soot emissions. Moreover, zero-dimensional single-zone combustion model is developed to compare with the experimental data.

2. Material and method

2.1. Experimental set-up

The experiments were carried out with a single cylinder, naturally aspirated, four-stroke, and water cooled, "Superstar" DI Diesel engine with a bowl in piston combustion chamber. Table 1 and Fig. 1 show the engine specification and experimental set-up, respectively.

So as to measure brake torque, the engine is coupled with a "Baturalp–Taylan" brand hydraulic type dynamometer of 50 kW absorbing capacity using an "S" type load cell with the precision of 0.1 N. Before starting the experiments, load cell is calibrated sensitively.

In this study, MRU Spectra 1600 L type gas analyzer was used so as to measure exhausts. Before experiments, emission and smoke device were calibrated.

Table 1
Engine specification.

Engine type	Super star
Bore [mm]	108
Stroke [mm]	100
Cylinder Number	1
Stroke Volume [dm^3]	0.92
Power, 1500 rpm [kW]	13
Injection pressure [bar]	175
Injection timing [crank angle]	35
Compression ratio	17
Maximum speed [rpm]	2500
Cooling	Water
Injection	Direct injection

In order to inject the steam into the engine, electronically controlled steam injection system was developed. Steam injection system is shown in Fig. 2.

In order to measure inside cylinder pressure, Kistler brand 6061B model, water cooled piezo-electric sensor and Kistler 5018 type charge amplifier was used in single cylinder engine. Smetec brand four channel data card which has 1 Mbyte data signaling rate from single channel "Combi Combustion Indication System" was used for data transfer, and Koyo TRD J1000-RZ type encoder which has 1000 pulse/revolution was used in order to measure angular position.

Experiments were done in variable speeds 1200, 1400, 1600, 1800, 2000, 2200 and 2400 rpm and full load condition.

In order to compare, firstly standard diesel tests were carried out. Then, saturated liquid water in a condition with 3 bar pressure and 133.5 °C temperature was injected with throttling (constant enthalpy) in intake period to intake manifold via injector. Steam amount was determined with a mass ratio of injected fuel. In the experiments, 10%, 20% and 30% steam ratios were used. The experiments were repeated for each steam ratios performance and emission values obtained were compared with those of standard diesel.

2.2. Theoretical model

In the literature, various combustion models were used to estimate the parameters of engine performance and emissions. Qi et al. predicted the effective power and SFC, NO and soot emissions with 2.8% and 9.1% errors with quasi-dimensional combustion model [23]. Sahin and Durgun developed a multi-zone combustion model to estimate the parameters of the diesel engine cycles and performance. In the study, effective power and SFC were calculated with 11–18% error when compared the experimental results [24]. Kannan et al. calculated in-cylinder pressures of diesel engine running ethanol and jatropha methyl ester with 1–2% errors by using two-zone combustion model [25]. Scappin et al. estimated SFC and NOx emissions with 95% accuracy by zero-dimensional two-zone combustion model [26]. In this study, thermodynamic simulation of a steam injected diesel engine carried out in terms of effective power, in-cylinder temperature and pressure, effective efficiency, SFC, torque and NO emissions by using zero-dimensional single-zone combustion model.

In theoretical model, it is assumed that gas mixture is homogeneous in the engine cylinder, thermodynamic properties, blow-by coefficient and residual gas fraction is constant and air–fuel mixture is ideal.

In the cylinder, the energy equations in differential form may be written as [27].

$$\frac{dU}{d\theta} = \frac{dQ}{d\theta} + \frac{dW}{d\theta} + \frac{dm_{fb}}{d\theta}h_f - \frac{dm_l}{d\theta}h_l \tag{1}$$

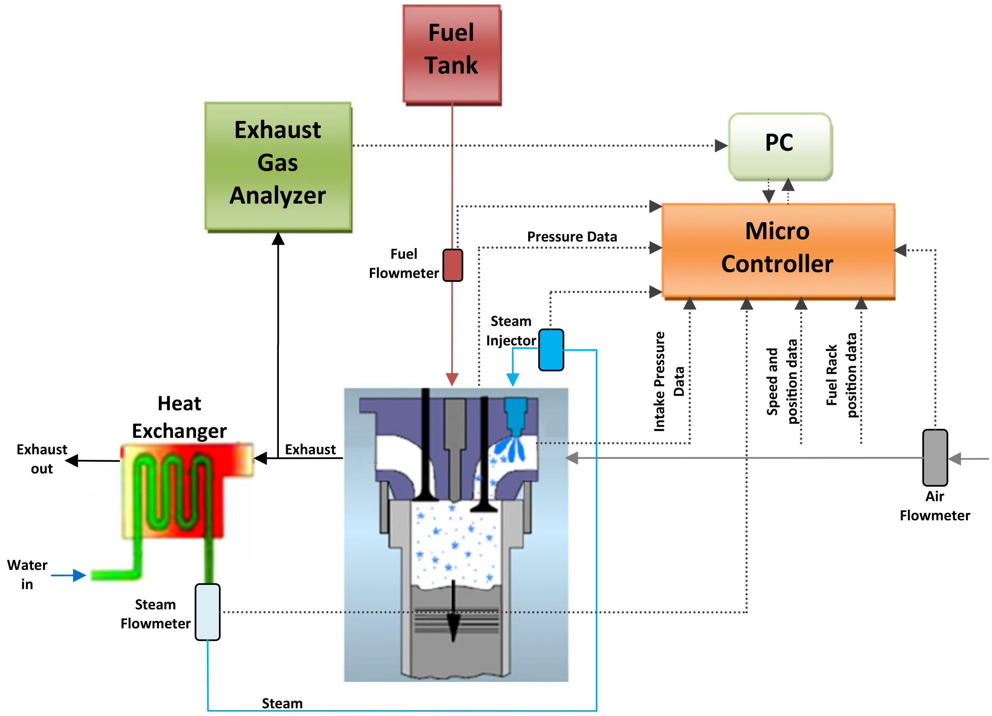

Fig. 1. Experimental set-up.

Fig. 2. Steam injection hole in cylinder head and steam injector in the hole.

where m_l is leak mass and m_{fb} is mass of burned fuel; h_f and h_l are enthalpies of burned fuel and leak mass, respectively. The mass rate of burned fuel can be stated as:

$$\frac{dm_{fb}}{d\theta} = \frac{\dot{m}_{fb}}{\omega} \tag{2}$$

where $\dot{m}_{fb}$ is the crank angle-dependent burned fuel rate and it can also be given as:

$$\dot{m}_{fb} = \dot{x}_b m_f \tag{3}$$

$$\frac{m_{fb}}{d\theta} = \frac{x_b}{d\theta} m_f \tag{4}$$

where m_a, m_f and x_b are masses of the air, the total mass of the injected fuel and fraction rate of the total burned fuel mass, respectively and F_{st} and RGF are stoichiometric fuel air ratio and residual gas fraction, respectively which can be written as:

$$m_f = \phi F_{st}(1 - RGF)m_a \tag{5}$$

The combustion progress variable x_b is estimated according to Vibe model. a_v and m_v are Vibe energy converting factor and Vibe Form Factor, respectively.

$$x_b = 1 - e^{-a_v\left(\frac{\theta-\theta_s}{\theta_b}\right)^{(m_v+1)}} \tag{6}$$

$$\frac{dx_b}{d\theta} = a_v\left(\frac{\theta-\theta_s}{\theta_b}\right)^{m_v}\left(\frac{m_v+1}{\theta_b}\right)e^{-a_v\left(\frac{\theta-\theta_s}{\theta_b}\right)^{(m_v+1)}} \tag{7}$$

Dual Vibe function express the burn fraction and x versus CA issued to state the heat release from combustion [28,29]. It is determined as [29]:

$$x = a_v\left[Q_{pre}\left(1 - e^{-a_v\left(\frac{\theta}{\theta_{pre}}\right)^{(m_{pre}+1)}}\right) + Q_{dif}\left(1 - e^{-a_v\left(\frac{\theta}{\theta_{dif}}\right)^{(m_{dif}+1)}}\right)\right]$$

where Q_{pre} and Q_{dif} are heat release rate of premixed and diffusion combustion, respectively and can be rewritten by differentiating with respect to crank angle.

$$\frac{dx}{d\theta} = a_v\left[\frac{Q_{pre}}{\theta_{pre}}(m_{pre}+1)\left(\frac{\theta}{\theta_{pre}}\right)^{m_{pre}} e^{-a_v\left(\frac{\theta}{\theta_{pre}}\right)^{(m_{pre}+1)}} + \frac{Q_{dif}}{\theta_{dif}}\left(m_{dif}+1\right)\left(\frac{\theta}{\theta_{dif}}\right)^{m_{dif}} e^{-a_v\left(\frac{\theta}{\theta_{dif}}\right)^{(m_{dif}+1)}}\right] \tag{8}$$

$$\theta = \theta_r - \theta_b \tag{9}$$

where θ_r and θ_b are reference crank angle and start angle of combustion respectively and a_v, m_{pre}, θ_{pre}, m_{dif}, θ_{dif} are Vibe constants in the premixed and diffusive combustion conditions [29].

x_b is given as burning fraction. It is 0 at the start of combustion and it would be 1 at the end of combustion. It is determined according to experimental results. θ, θ_s and θ_b are instant crank angle, crank angle at the start of burning, burning duration in crank angle, respectively.

The enthalpy of burned fuel could be expressed as:

$$h_f = \Delta h/M_f + (P_i - 1.01325)v_f 10 \tag{10}$$

where Δh, M_f, P_i and v_f are combustion enthalpy, molecular weight, injection pressure and specific volume of the fuel, respectively. The heat passed through the system boundary with respect to crank angle is given as:

$$\frac{dQ}{d\theta} = -\frac{\dot{Q}_l}{\omega} \tag{11}$$

where ω is angular velocity and the heat loss rate can be obtained as follows:

$$\dot{Q}_l = h_{tr}A_{cyl}(T - T_w) \tag{12}$$

where h_{tr}, A_{cyl}, T and T_w are heat transfer coefficient, heat transfer area of the cylinder, temperatures of the in-cylinder gas zone and cylinder walls, respectively [27]. The heat transfer coefficient (h_{tr}) is found by using Hohenberg's [30] approach and written as below:

$$h_{tr} = C_1 V^{-0.06} P^{0.8} T^{-0.4}\left(\bar{S}_p + C_2\right)^{0.8} \tag{13}$$

where $C_1 = 130$, $C_2 = 1.4$ and $\bar{S}_p$ is mean piston velocity in meters per second, respectively. The crank angle-dependent statement of the work is given as:

$$\frac{dW}{d\theta} = -p\frac{dV}{d\theta} \tag{14}$$

where p and V are in-cylinder pressure and volume, respectively. The change of stroke volume depending on crank angle is following:

$$\frac{dV}{d\theta} = \frac{\pi}{8}B^2 S\sin\theta\left[1 + \varepsilon\frac{\cos\theta}{(1-\varepsilon^2\sin^2\theta)^{\frac{1}{2}}}\right] \tag{15}$$

where B, S and ε are bore, stroke of the cylinder and the ratio of half stroke to rod length, respectively. The time (crank angle)-dependent burned gas leaking blow-by through the rings is:

$$\frac{dm_l}{d\theta} = \frac{Cm}{\omega} \tag{16}$$

In this study, C governs the loss of gas mass (blow-by). Mass balance within the cylinder can be expressed as follows:

$$m = m_a + m_{fi} \tag{17}$$

where m_a and m_{fi} are the masses of the air and injected fuel respectively. If the Eq. (12) is written in differential form, it becomes as follows:

$$\frac{dm}{d\theta} = \frac{dm_a}{d\theta} + \frac{dm_{fi}}{d\theta}. \tag{18}$$

The air and injected fuel rates changing with crank angle within the cylinder are expressed respectively as:

$$\frac{dm_a}{d\theta} = \frac{-\dot{m}_l/\omega}{1+\phi F_{st}} = \frac{-Cm_a}{\omega} \tag{19}$$

$$\frac{dm_{fi}}{d\theta} = \frac{1}{\omega}\left(\dot{m}_{fi} - \frac{\dot{m}_l \phi F_{st}}{1+\phi F_{st}}\right) = \frac{\dot{m}_{fi} - Cm_{fi}}{\omega} \tag{20}$$

where $\dot{m}_l$, ϕ and F_{st} are the time-dependent gas leak rate, the equivalence ratio and the stoichiometric fuel–air ratio by mass, respectively. $\dot{m}_{fi}$ is the time-dependent injected fuel rate and it may be stated as:

$$\dot{m}_{fi} = \dot{x}_i m_f \tag{21}$$

where $\dot{x}_i$ is fraction rate of the total injected fuel mass, which can be given as:

$$\dot{x}_i = \frac{\omega}{\theta_{di}\Gamma(n)}\left(\frac{\theta-\theta_{si}}{\theta_{di}}\right)^{n-1} \exp\left[\frac{-(\theta-\theta_{si})}{\theta_{di}}\right] \tag{22}$$

where θ_{di} is a parameter of injection duration, θ_{si} is the start of fuel injection, $\Gamma(n)$ is the gamma function [27], θ_{db} is a parameter of burning duration, θ_{sb} is the start of burning, RGF is residual gas fraction. The gamma function is derived as:

$$\ln\Gamma(n) = \left(n-\frac{1}{2}\right)\ln(n) - n + \frac{1}{2}\ln(2\pi) + \frac{1}{12n} - \frac{1}{360n^3} + \frac{1}{1260n^5} - \frac{1}{1680n^7} \tag{23}$$

The value of n could be taken for the diesel engine with open chamber as $1 \le n \le 2$ and for close chamber as $3 \le n \le 5$. But exact value is dependent on fuel used and engine design [27].

The variation of c_p with temperature is taken into consideration with respect to Janaf tables which is given by Ferguson et al. [27]. The time (crank angle)-dependent expressions of pressure and mean gas temperatures are given respectively as:

$$\frac{dp}{d\theta} = \frac{-\frac{pV}{T}\left[\frac{10\,c_pT}{pV} - \frac{\partial \ln v}{\partial \ln T}\right]\frac{dv}{d\theta} - \frac{10\frac{du}{d\theta}\frac{\partial \ln v}{\partial \ln T}}{p}}{\frac{V^2}{T}\left[\left(-\frac{10\,c_pT}{pV} + \frac{\partial \ln v}{\partial \ln T}\right)\left(-\frac{\partial \ln v}{\partial \ln p}\right) + \frac{\partial \ln v}{\partial \ln T}\left(\frac{\partial \ln v}{\partial \ln T} + \frac{\partial \ln v}{\partial \ln p}\right)\right]} \tag{24}$$

$$\frac{dT}{d\theta} = \frac{-V\left[\frac{10\left(\frac{du}{d\theta}\right)}{p\frac{\partial \ln v}{\partial \ln P}} + \left(\frac{dv}{d\theta}\right)\left(\frac{\partial \ln v}{\partial \ln T} + \frac{\partial \ln v}{\partial \ln p}\right)\right]}{\frac{V^2}{T}\left[\left(-\frac{10\,c_pT}{pV} + \frac{\partial \ln v}{\partial \ln T}\right)\left(-\frac{\partial \ln v}{\partial \ln p}\right) + \frac{\partial \ln v}{\partial \ln T}\left(\frac{\partial \ln v}{\partial \ln T} + \frac{\partial \ln v}{\partial \ln p}\right)\right]} \tag{25}$$

In order to solve the differential equations given above, the modified RATES and STATE codes [27] developed by Gonca [31] are used. The effective power and thermal efficiency are expressed as:

$$P_e = \frac{W_e N}{120} \tag{26}$$

$$\eta_e = \frac{P_e}{\dot{m}_f H_u}. \tag{27}$$

NO emissions are calculated by using extended Zeldovich mechanism taking into account 10 combustion products including (CO_2, H_2O, N_2,O_2, CO, H_2, H, O, OH, NO) [32]. In this study the ECP code which is developed by Olikara and Borman [33] is modified by adding steam injection into the reactants. The combustion reaction used in the modified program is given below:

$$\varepsilon\phi C_\alpha H_\beta O_\gamma N_\delta + (0.21O_2 + 0.79N_2) + xH_2O \rightarrow \nu_1 CO_2 + \nu_2 H_2O + \nu_3 N_2 + \nu_4 O_2 + \nu_5 CO + \nu_6 H_2 + \nu_7 H + \nu_8 O + \nu_9 OH + \nu_{10} NO$$

The x in the reactants is mole fraction of injected steam and can be calculated as:

$$x = \frac{Y_{\%} M_f}{M_{ste}} \tag{28}$$

where M_f and M_{ste} are molecular weights of the fuel and steam respectively. $Y_{\%}$ is ratio of the steam mass to the fuel mass and defined as:

$$Y_{\%} = \frac{m_{ste}}{m_f} \tag{29}$$

The rate constant is expressed as:

$$k = A_A T^{B_A} e^{\frac{E_A}{T}} \tag{30}$$

The rate of NO formation [mol cm^{-3} s^{-1}] is [32];

$$\frac{d[NO]}{dt} = \frac{2R_1(1-\alpha^2)}{1+\frac{\alpha R_1}{R_2+R_3}} \tag{31}$$

where $\alpha = [NO]/[NO]_e$ and $[\ ]_e$ denotes equilibrium concentration. The other constants used in Eq. (31) are:

$$R_1 = k_{+1}[N_2]_e[O_2]_e = k_{-1}[NO]_e[N]_e \tag{32}$$

$$R_2 = k_{+2}[O_2]_e[N]_e = k_{-2}[NO]_e[O]_e \tag{33}$$

$$R_3 = k_{+3}[OH]_e[N]_e = k_{-3}[NO]_e[H]_e \tag{34}$$

3. Results and discussion

In this study, zero-dimensional single-zone combustion model is used to investigate the effects of electronically controlled steam injection on performance and emission values of a diesel engine. A theoretical model is developed for the optimum steam rate (20%) determined by experimental data [20,21] and the model is compared with experimental data.

3.1. Performance parameters

3.1.1. Cylinder pressure

The comparison of in-cylinder pressures of theoretical and experimental data at full load conditions is indicated in Fig. 3.

It can be seen from Fig. 3 that compression and expansion work in experimental S20 reduce in comparison to theoretical S20. However, when net total work is taking account, net work obtained with steam injection (S20) is more than with standard diesel engine in experimental study.

3.1.2. Heat release

The comparison of heat release of theoretical and experimental data at full load conditions is indicated in Fig. 4. The figure shows

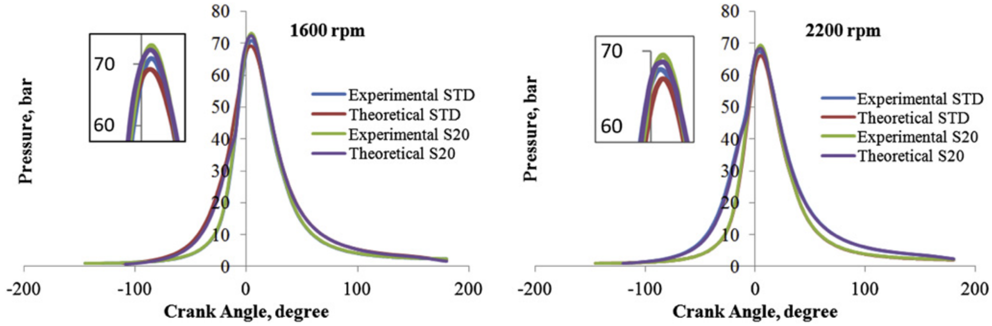

Fig. 3. Comparison of theoretical and experimental data of cylinder pressure.

the double-peak shape which characterizes conventional combustion profile of a direct injection diesel engine. The first peak in the figure is called as premixed combustion, which strongly depends on the amount of fuel prepared for combustion during the ignition delay period. The second peak is diffusion combustion which is controlled by the fuel–air mixing rate and diffusion combustion continues until combustion is completed [34]. As can be seen from the figure, the heat release rate increases both premixture and diffusion phases until 5 CA after TDC for the S20 comparing with standard diesel. Heat release rate begins to decrease leading to reduce in-cylinder gas temperature. Thus, NO emissions reduce due to lower temperature and heat release after that period. It can be concluded that as maximum pressure has been reached before 5 CA after TDC comparing to standard diesel. Effective power and torque increased compared to standard diesel.

3.2. Engine effective parameters

3.2.1. Engine torque

Fig. 5 shows the comparison of theoretical and experimental data of brake torques. As can be seen from the figure, steam injection improves the engine torque.

According to experimental results, torque increases at all engine speeds with 20% steam injection rate (S20). Maximum torque attained is found as 60.2 Nm at 1600 rpm. While the highest increase in torque is 2.5% at 1200 rpm; the lowest improvement is 1.2% at 1600 rpm.

In Fig. 3 (extended figure on left side), pressure data obtained from experimental study along with 10 Crank Angle (CA) after Top Dead Center (TDC), in-cylinder pressure with S20 experimental study is more than with S20 theoretical study. Thus, in this range of higher pressure is resulted in the higher torque in experimental S20.

3.2.2. Effective power

Fig. 6 shows the comparison of the theoretical and experimental results of effective power. As it is observed in the brake torque data, effective power also increases at all engine speeds with S20. The maximum effective power is 13.3 kW at 2400 rpm. According to standard torque values, the highest change is 2.5% at 1200 rpm, the lowest change is 1.2% at 1600 rpm.

3.2.3. Specific fuel consumption and effective efficiency

In Fig. 7, theoretical and experimental data of specific fuel consumption (SFC) is given comparatively. As can be seen from the figure that SFC reduces at all engine speeds with S20. The lowest SFC is 263.7 g/kWh at 1600 rpm. According to standard SFC, the highest change is 6.1% at 2400 rpm, the lowest change is 0.4% at 1200 rpm. Fig. 8 gives the theoretical and experimental data of effective efficiency. The improvement rates of effective efficiency are same as SFC.

The most possible reason to reduce in SFC and increase in torque, effective power and efficiency by means of steam injection could be explained with the improvement in vaporization and mixing processes which leads to a shorter combustion reaction [20,35]. Moreover, the improved SFC with S20 could be explained owing to the presence of the diesel oil–water steam interface with very low interfacial tension which leads to a finer atomization of the fuel during injection. Higher contact with the air during the burning process is resulted in a finer dispersion of the fuel drops. Depression of thermal dissociation could be the third reason for the improved combustion efficiency [20,21].

3.3. Emission parameters

3.3.1. In-cylinder temperature and NO formation

The comparison of in-cylinder temperatures of theoretical and experimental data at full load conditions for the engine speed rates

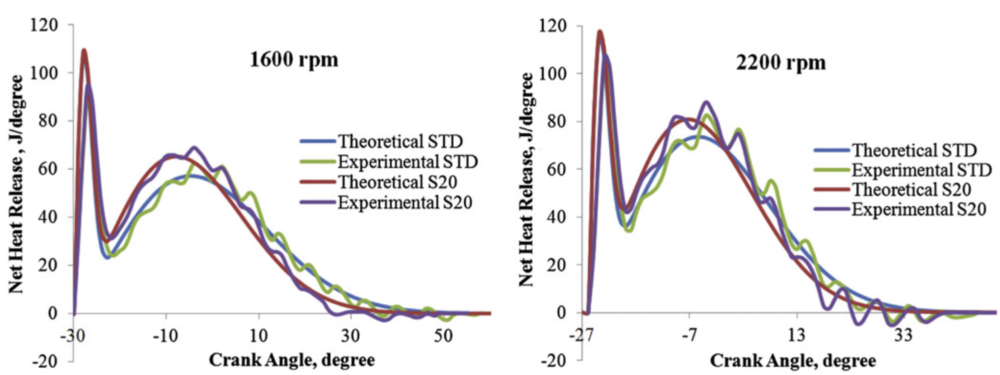

Fig. 4. Comparison of theoretical and experimental data of net heat release.

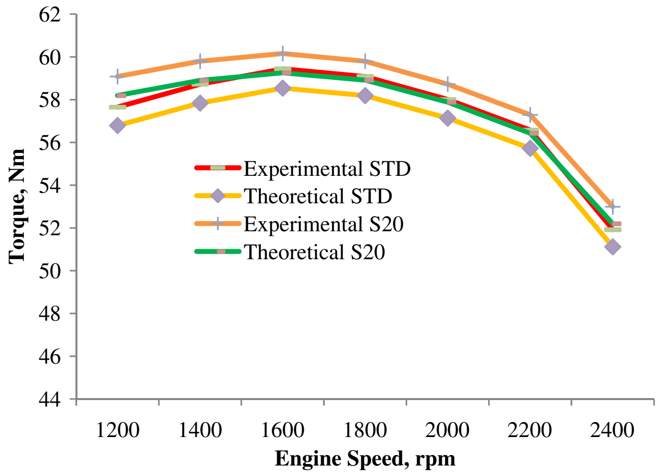

Fig. 5. Comparison of theoretical and experimental data of torque.

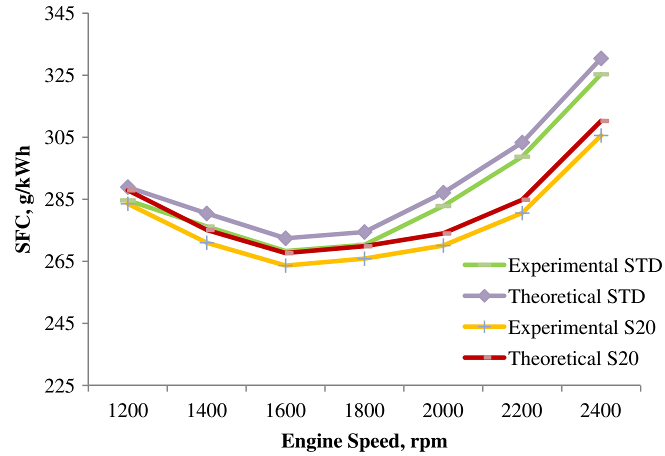

Fig. 7. Comparison of theoretical and experimental data of SFC.

tested is given in Fig. 9. In-cylinder temperatures are determined by substituting in Eq. (25) via measuring pressure data with pressure sensor. These values are compared with theoretical temperature obtained by means of pressure values in Eq. (24). The theoretical simulation has good agreement with experimental results. The effects of steam injection on lowering in-cylinder gas temperature are clearly shown in the figure.

It is well known that NO formation rate strongly depends on peak temperature and duration of combustion at peak temperature in the cylinder [32]. Thus, when the steam injection is performed, peak temperatures decreased compared to that of standard diesel engine.

Fig. 10 gives the comparison of theoretical and experimental data of NO emissions. As can be seen from the figure, steam injection caused to decrease NO emissions as the peak temperatures decrease.

The NO emissions decrease at all engine speeds with optimum steam injection rate. The minimum NO is 475 ppm at 2400 rpm. According to standard NO emission values, the highest change is 22.4% at 1200 rpm, the lowest change is 7.4% at 2200 rpm.

The interest in water injection techniques are due to the fact that water in the form of micrometer sized droplets exerts some positive effects on the combustion of the fuel and exhaust emissions, frequently NO*x*. Water injection into diesel engines shows some interesting effects, such as reduced NO*x* and improved combustion efficiency. The finely atomized water droplets vaporize instantly after being injected into the combustion chamber. The combined effect of vaporization absorbing heat, relatively high molar heat capacity of water and increased partial pressure of oxygen puts down the peak combustion temperature and thus decreases the nitrogen oxides formation [36,37]. Fig. 10 shows the effect of water steam injection on NO emissions. As can be seen from the figure, NO emissions reduced with all the engine speeds with 20% steam.

3.3.2. *CO and CO_2*

Fig. 11 compares CO emissions of steam injected diesel engine with the standard one. The CO emissions enhance with optimum steam injection rate at all engine speeds. However, it is seen that the increase in the emission is within the limits of uncertainty when considering measurement accuracy. The minimum CO emission is 0.35% at 2400 rpm in standard condition.

The comparison of CO_2 emissions of steam injected and standard diesel engine is shown in Fig. 12. The CO_2 emissions reduce with steam injection at all the engine speeds. The minimum CO_2 emission is 10.2% at 2400 rpm at steam injected condition. The maximum reduction in CO_2 emissions is 4.3% at 1800 rpm. As a result, CO_2 emissions reduced because SFC decreased with the steam injection as can be seen from Fig. 7.

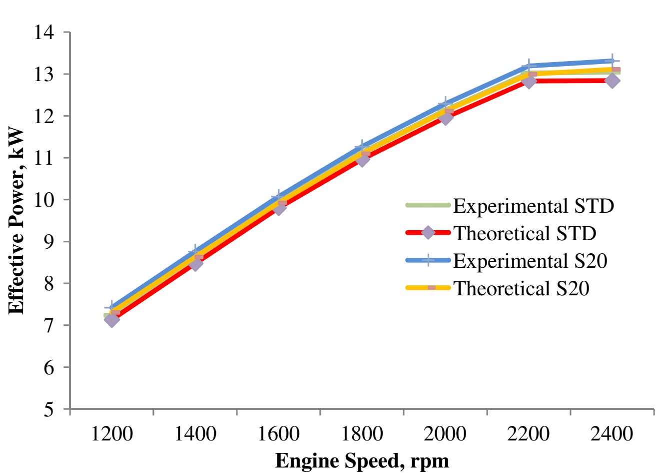

Fig. 6. Comparison of theoretical and experimental data of effective power.

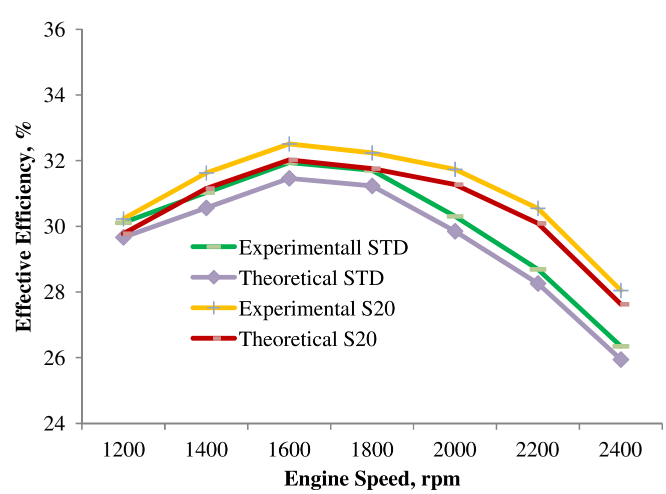

Fig. 8. Comparison of theoretical and experimental data of effective efficiency.

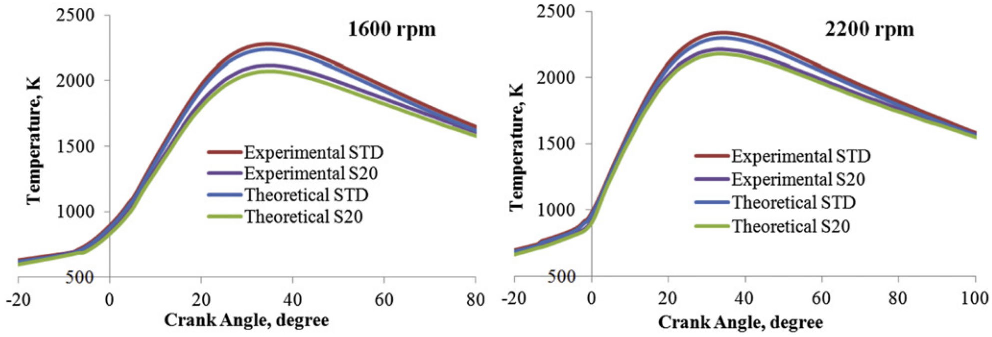

Fig. 9. Comparison of theoretical and experimental data of cylinder temperature.

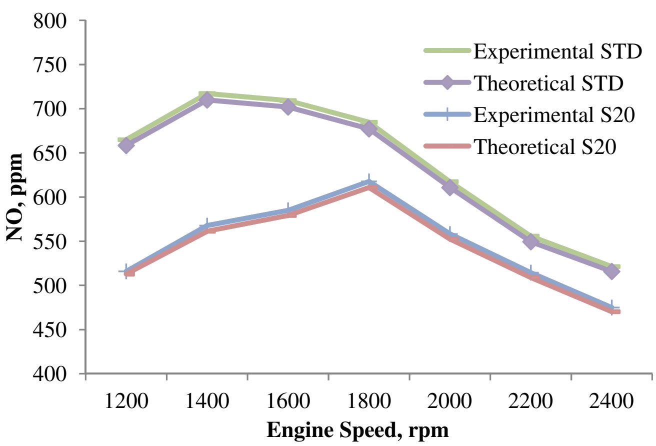

Fig. 10. Comparison of theoretical and experimental data of NO emissions.

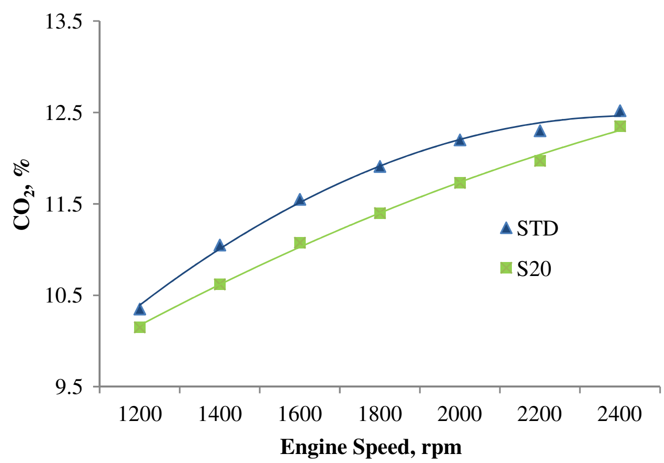

Fig. 12. Comparison of CO_2 emissions of standard and steam injected diesel engines.

3.3.3. HC

Fig. 13 illustrates the comparison of HC emissions of steam injected and standard diesel. The HC emissions increase with steam injection until 2000 rpm. However, it is seen that the changes in the emission values of standard and steam injected diesel engines are within the limits of uncertainty when considering measurement accuracy. The minimum HC emission is 5.5 rpm at 2400 rpm of 20% steam injection.

3.3.4. Smoke density

The smoke density of steam injected and standard diesel engine is compared in Fig. 14. It is not observed a remarkable change in the smoke density with steam injection. The increase in the emission is within the limits of uncertainty when considering measurement accuracy. The value of smoke density is raised from 44% to 46% at 2200 rpm for steam injection condition.

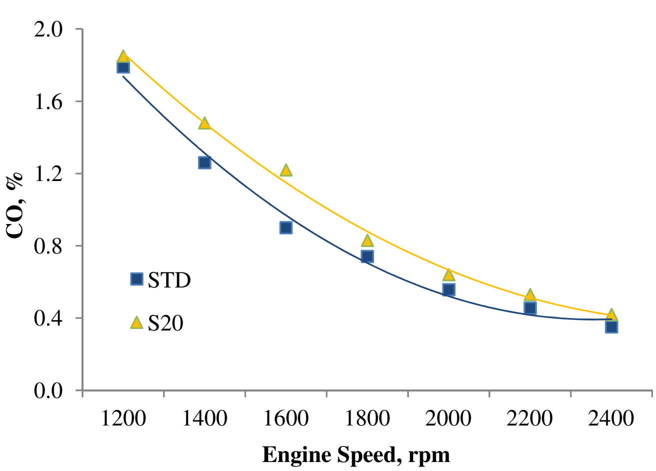

Fig. 11. Comparison of CO emissions of standard and steam injected diesel engines.

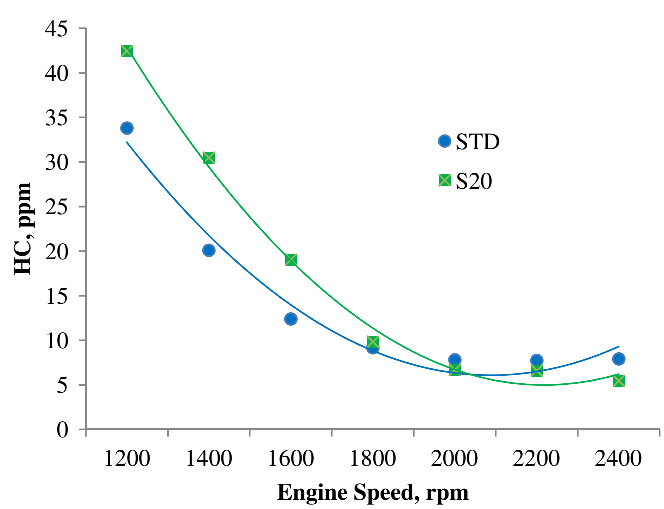

Fig. 13. Comparison of HC emissions of standard and steam injected diesel engines.

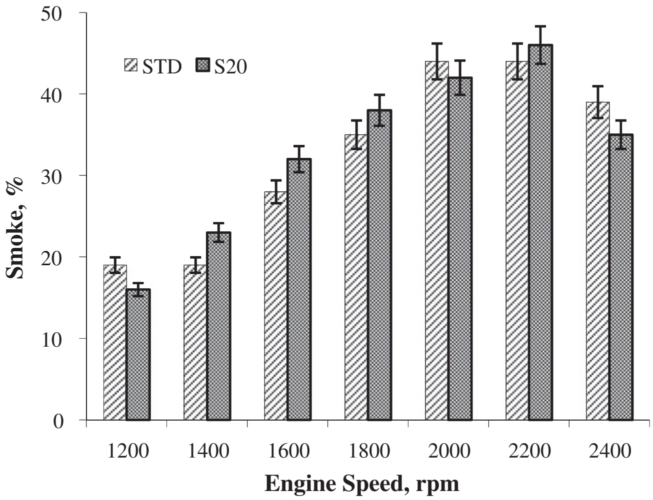

Fig. 14. Comparison of smoke density of standard and steam injected diesel engines.

4. Conclusion

In this study, the effects of electronically controlled steam injection system developed by Parlak et al. [20,21] on performance and emissions have been investigated and modeled by zero-dimensional single-zone combustion model. When compared with experimental and theoretical data, torque, effective power and efficiency, SFC and NO emissions are closed to actual values with 1.5% maximum error.

There is an improvement in torque, effective power and efficiency and SFC with steam injection and NO emissions decrease significantly. The maximum torque was obtained at 1600 rpm with optimum rate of 20% steam injection. The highest change in torque and power is determined at 1200 rpm as 2.5%. Furthermore, maximum power is obtained at 2400 rpm. Whilst minimum SFC and maximum effective efficiency are obtained at 1600 rpm, the highest change as a standard value of SFC and effective efficiency is at 2400 rpm. The minimum NO emission value is determined at 2400 rpm. The highest change with steam injection is at 1200 rpm.

Consequently, steam injection yields positive effect on performance and NO emissions at all speeds. It is clear that steam injection does not affect CO, CO_2, HC emissions considerably.

Thus, this study could be implemented to the engineering applications; it may be a leading essential study for the real-engine designers by considering the effects of steam injection into the engine cylinder.

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